

Fuel and emissions properties of Stirling engine operated with wood powder

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Abstract

From viewpoints of the environment and fuel cost reduction, small-scale biomass combined heat and power (CHP) plants are in demand, especially wood-waste fueled system, which are simple to operate and maintenance-free with high thermal efficiency similar to oil fired units. These are requested by wood and other industries located in mountainous region. To meet these requirements, a Stirling engine CHP system combined with simplified biomass combustion process with pulverized wood powder was developed.

In an R&D project started in 2004 considering wood powder properties as a fuel, combustion performance and emissions in combustion flue gas were tested using combustion test apparatus with commercial size units. The wood powder combustion system was modified and optimized during the combustion test results, and the design of the demonstration plant combined with 55 kW_e Stirling engine power unit was considered. The demonstration plant was finally completed in March of 2006, and test operation has been progressed for the future commercial CHP system.

In the wood powder combustion test, wood powder of less than 500 μm is mainly used, and a combustion chamber length of 3 m is applied. In these conditions, the air ratio can be reduced to 1.1 without increasing CO emission of less than 10 ppm and combustion efficiency of 99.9%. In the same conditions, NO_x emission is estimated to be less than 120 ppm (6% O₂ basis). Wood powder was confirmed to have excellent properties as a fuel for Stirling engine CHP system. This paper summarizes the wood powder combustion test, and presents the evaluation of the burner design parameters for the biomass Stirling engine system.

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1. Introduction

From the environmental point of view, the use of biomass will help establish a sustainable society, because biomass has the advantages of being a renewable energy source and is neutral on carbon dioxide production. Development

and commercial installation with the new power technologies utilizing biofuel have aggressively progressed in Japan [1]. The use of biomass in a large scaled power plant has also begun. The Japanese RPS (Renewable Portfolio Standard) program changed the electric power companies and the Independent Power Producers, and they positively promote the application of biofuel. Wood biomass co-firing with coal in a large scaled pulverized coal (PC) fired power plant is one of the most promising processes and many demonstration projects are under way

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[2]. This process can use much of the equipment of the existing power plant and it can expect to have quite high thermal efficiency utilizing biomass [3]. If the costs of biomass collection and its transportation can be made lower than the cost of coal, this process can be expected to spread greatly [4]. Biomass co-firing with oil fuel is also under development, and this technology is expected to not only result in CO₂ reduction but also result in oil fuel cost saving given the current oil cost situation [5].

In mountainous regions, there is a large amount of wood waste from the forestry and wood industries. However, Japanese forestry has atrophied and domestic forest devastation is spreading [6]. Because of this situation, much of wood-waste discharging in mountainous regions is left in the forests of Japan [7]. Furthermore, the rapid rise of oil prices has impacted the wood and other industries, and has initiated the shift from oil as the primary energy resource to woody waste biomass. Therefore, a distributed power system and small-scaled CHP (combined heat and power) plant with biofuel is applicable for the market of Japanese mountainous regions. Many suppliers are developing small-scale biomass gasification CHP systems to meet these requirements in Japan, and many types of biomass gasifiers are proposed and developed aggressively [8]. We also studied a usable gasification process of small scale, and as a result, we would assure that these systems are simple to operate and are maintenance free with high thermal efficiency similar to oil fired units [9].

To satisfy these requirements, an external combustion engine such as the Stirling engine is expected for the small-scaled power generation system with high thermal efficiency, and it can operate with high temperature fuel gas. This means that the biomass energy conversion process applied to the Stirling engine is more flexible and simpler than conventional biomass gasification with a gas engine [10]. On the other hand, wood powder burner could also be used in a simple combustion system, because wood powder has many advantages as burner fuel such as high heating value, high bulk density, small fuel size, good flow ability and others [10]. We started to develop a Stirling engine CHP system applying a simplified biomass combustion process with a pulverized wood powder burner instead of a gasifier CHP system with gas engine in 2004.

A package unit of 55 kW_e Stirling engine unit using clean gas fuel started the commercial selling in 2005. We selected this power unit for our CHP system development [11]. In addition to this Stirling engine, development of the biofuel Stirling engine has progressed aggressively in Europe and other countries. A wood gas fuelled Stirling engine with 9 kW_e power generation has been successfully run in Denmark [12]. 35 kW_e Stirling engine CHP system assembled with wood chip fired boiler has been operation since 2002 in Austria [13], and a 75 kW_e Stirling engine pilot plant with eight-cylinders designed based on the above experiences has been in operation since 2003 [14]. A wood powder combustion project with the same 35 kW_e Stirling engine has also started in Sweden [15].

Furthermore, new technologies of solar power Stirling engine [16] and low temperature Stirling engine [17] with further small units of 10 kW_e are also advanced. Stirling engines can be expected as a promising technology for the renewable energy utilization.

Many experiences on the wood powder burning technology of co-firing with coal or oil are demonstrated as mentioned above. And fuel properties, combustion and emissions performances and risks of bio-fuel are also reported [18,19]. However, the development of the wood powder burner was the key issue when this process was established because of the wood powder burner of 100% biofuel. Therefore, combustion tests of wood powder were carried out from the beginning of 2005 (STEP-I). In the next stage (STEP-II), the dummy heater test rig used to simulate the actual Stirling engine was installed at the exit of the combustion chamber, because we have been feeling the risk of ash deposits in heater tubes as reported [20,21]. As a result of STEP-II test, since the ash deposits in heater tubes and fins were clearly found, the trial tests for solving or reducing the ash deposits are being carried out with actual heater tubes and fins of 55 kW_e Stirling engine. These results will be reported in the future. System demonstration tests will be carried out through March 2007 (STEP-III). This paper presents combustion test results in STEP-I, which is a key technology of this system. Also discussed are the combustion parameters for the Stirling engine power system design.

2. Experiment and methodology

2.1. Test apparatus

The general specification of the test apparatus is shown in Table 1. A Stirling engine unit of 55 kW_e power output is applied in STEP-III. A Stirling engine of clean gas fuel type is used for the test, and some Stirling engine parts were modified for wood powder combustion. The biomass burning capacity of the test apparatus is sufficient to operate this engine. The maximum burning capacity of the burner is 0.03 kg/s of wood powder, which is a thermal capacity of 400 kW. In case of the 55 kW_e CHP system, fuel consumption in normal operation is approximately 0.02 kg/s.

The test apparatus for wood powder combustion is shown in Fig. 1. Wood powder fuel is transported by air from the fuel hopper, and the fuel feed rate is controlled

Table 1
General specification of test apparatus

Item	Specification
(1) Wood powder feed rate	Max. 0.03 kg/s
(2) Burner thermal capacity	400 kW
(3) Burner type	Diffusion mixing, dual fuel coaxial burner
(4) Combustion chamber type	Horizontal furnace, refractory lining
(5) Combustion chamber size	1.0 m ID × 6 m L
(6) Draft control	Balanced draft
(7) Main fuel	Pulverized wood powder
(8) Start-up fuel	LP gas

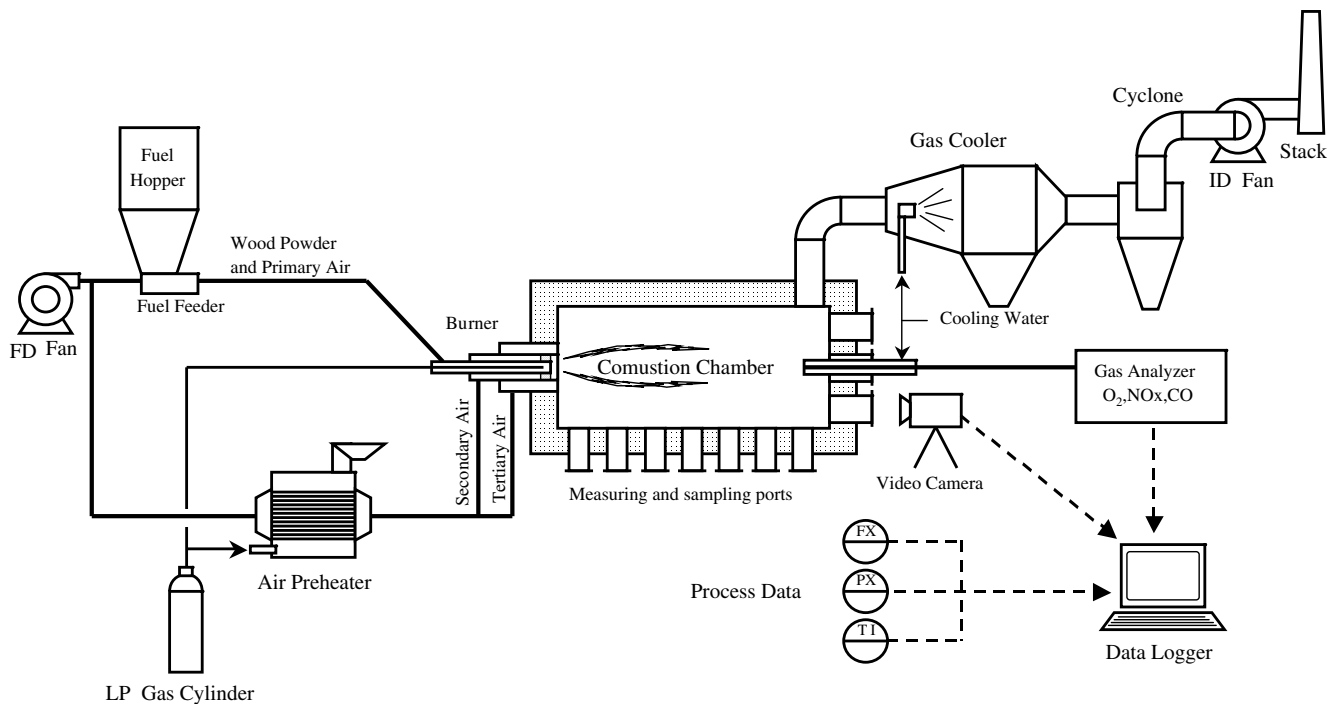


Fig. 1. Test apparatus of wood powder combustion.

by the rotating rate of the fuel feeder. Combustion air is supplied by a forced draft fan (FDF) and it is distributed to three air nozzles. There is primary air (PA) for fuel transportation, secondary air (SA) for main combustion air, and tertiary air (TA) for over firing air (OFA). SA is injected into the combustion chamber with a swirl, and TA is fed from outside of the SA without a swirl. PA is not pre-heated, and both SA and TA are preheated up to 300 °C with an air preheater fired with an LP gas burner. The tested burner is a dual, coaxial type burner, and is modified from a pulverized coal (PC) burner. The two types of burners shown in Table 2 were tested. Burner type A is a short flame burner to make high combustion efficiency a priority. Burner type B is a relatively long flame burner to reduce NO_x emission. The two burners were compared based on combustion and emission performance.

A schematic drawing of the wood powder burner and the combustion chamber is shown in Fig. 2. The original combustion chamber is 6 m long horizontally with refractory lining as shown in Fig. 2. The length of the combustion chamber can be changed easily. At the beginning of combustion test, the chamber length was 6 m. In test STEP-II and STEP-III, the length was reduced to 3 m and combustion performance was tested and compared with that of 6 m. After evaluation, the chamber length of 3 m was selected for the test STEP-III.

During cold start-up, LP gas is used for the ignition and start-up period, and the burner fuel is switched to wood powder when the chamber temperature reaches 800 °C. The exit gas temperature from the combustion chamber finally rises to 1200 °C by wood powder burning. Exhaust gas is cooled to less than 200 °C by water spray injection.

Table 2
Details of wood power burner

Type name		Type-A	Type-B	Note
(1) Burner thermal capacity	kW	400	400	
(2) Fuel feed rate, max	kg/s	0.030	0.030	
(3) Fuel feed rate, nor.	kg/s	0.018	0.018	
(4) Primary air (PA) ^a flow	Nm ³ /s	0.028	0.028	
(5) Secondary air (SA) flow	Nm ³ /s	0.040	0.040	
(6) Tertiary air (TA) flow	Nm ³ /s	0.040	0.040	
(7) PA temperature	°C	30	30	
(8) SA & TA temperature	°C	300	300	
(9) Fluid of each burner tubes (with triple tube)				
– Inner tube		LP gas	Biomass	w/o swirl
– Middle tube		Biomass	LP gas	w/o swirl
– Outer tube		SA	SA	w/ swirl

^a Air for fuel transportation.

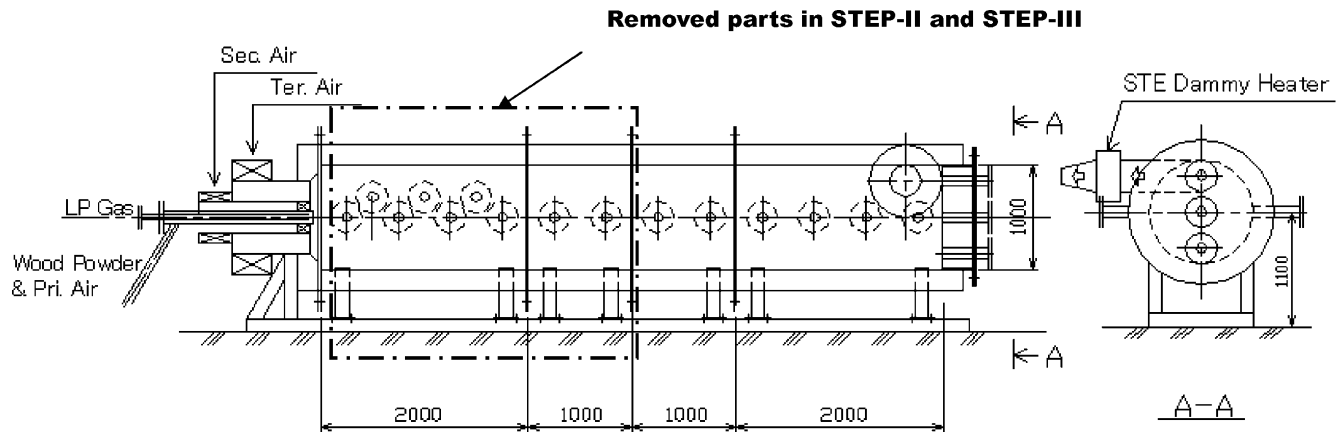


Fig. 2. Schematic diagram of combustion chamber.

Dust particles in the flue gas are removed by a single one-stage cyclone, and exhausted to the stack by an induced draft fan (IDF). The draft of the combustion chamber is controlled to zero by an ID fan with a draft control damper. An online monitoring and logging system is used for process and operation data. During operation, ash and flue gas for analysis are collected at the exit of the combustion chamber. Burner flame patterns are monitored by a video camera and by the operator's eye through the sidewall and rear wall.

2.2. Fuel analysis

Analysis of wood powder is shown in Table 3. Wood biomass as fuel is better than waste fuels and coals because of the following properties: high combustion efficiency due to high volatile matter [22,23], easy to control flue gas emissions due to low ash, nitrogen, and sulfur [24]. It is also possible to reduce erosion and corrosion risks because of low sulfur, chlorine and ash. However, fouling and slugging troubles in the heating surfaces have to be considered

Table 3
Fuel analysis

		Sample-1	Sample-2	Sample-3
Size	μm	<100	<250	<500
HHV	kJ/kg	21,220	21,030	21,080
<i>Proximate analysis</i>				
H ₂ O	%, wet	5.4	7.2	8.9
Volatile matter	%, dry	81.0	83.3	82.5
Fixed carbon	%, dry	18.6	16.4	17.2
Ash	%, dry	0.5	0.3	0.4
<i>Ultimate analysis</i>				
C	%, dry	53.3	52.7	53.0
H	%, dry	6.2	6.1	6.1
N	%, dry	0.2	0.1	0.1
S	%, dry	0.01	0.01	0.01
Cl	%, dry	0.01	0.01	0.01
Ash	%, dry	0.5	0.3	0.4
O	%, dry	39.8	40.7	40.3
Total	%, dry	100.0	100.0	100.0

due to high potassium. Therefore, the combustion test was carried out with dummy heater tubes in order to determine ash risks in the test period of STEP-II. The details of STEP-II are not included in this paper.

3. Results and discussion

3.1. Temperature profile of combustion chamber

Gas temperature in the combustion chamber was measured 100 mm from the chamber wall. The temperature distribution profiles in cases of burner types A and B with 6 m length and burner type B with 3 m length are shown in Fig. 3. In the case of a chamber length of 3 m, both average gas temperature and chamber outlet gas temperature were the highest in all three cases. These results can be estimated due to the smallest chamber wall surface area and the smallest radiation loss from chamber surface. The temperature peak points of the three cases are almost the same location of 1,500 mm from the burner. However, the temperature levels at peak point are very different between burner types A and B, and those are not independent of the chamber length. The temperature profile in the combustion chamber is very different between burner types A and B, but the outlet gas temperatures of the combustion chambers are almost the same.

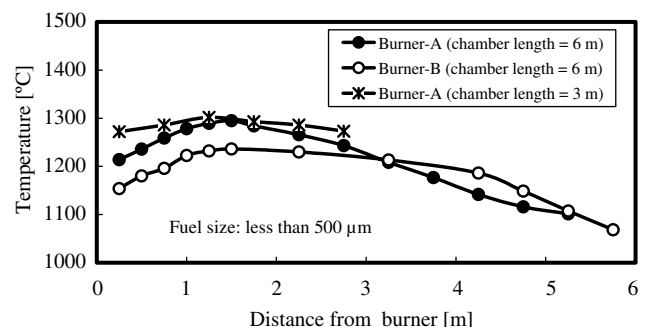


Fig. 3. Temperature distribution profile in combustion chamber.

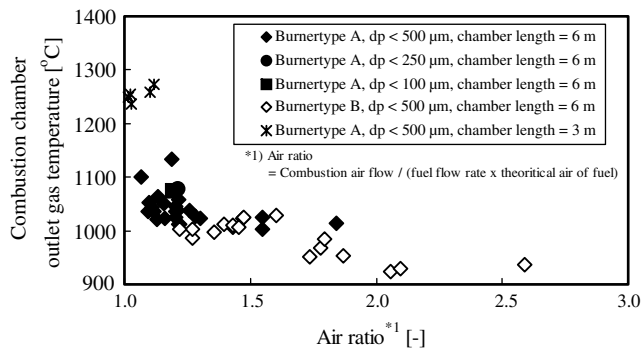


Fig. 4. Effect of air ratio on combustion chamber outlet gas temperature.

The results of temperature profile and peak temperature level can be explained by the correlation with burner flame length. The outlet gas temperature must be dependent on the combustion chamber length. With regard to the Stirling engine performance, higher gas temperature at the combustion chamber outlet should be expected to take the larger power output. Therefore, the shorter chamber length is better, proving that burner type A is better than burner type B. Additionally, NO_x emission from burner type A is estimated to be higher than that of type B. These are discussed in the following sections.

In general, burner flame temperature strongly correlates with air ratio. The relation between air ratio and combustion chamber outlet gas temperature is shown in Fig. 4. Here, “air ratio” is defined as the ratio of “actual combustion air flow rate” and “theoretical air flow rate of fuel”. All of the “air ratios” in this paper are defined in the same way. In case of higher air ratio, combustion chamber outlet gas temperatures are definitely lower, which is due to the cooling effect by air of 300 °C. As mentioned above, the average temperature of the 3 m chamber case is the highest in all cases, but the difference between burner-A and burner-B is not so clear. Furthermore, the effect of the fuel particle size on the temperature profile is also not clear. The relationship between air ratio and combustion chamber outlet gas temperature has a strong impact on the Stirling engine performance and is one of the most important parameters in this R&D program. This issue is discussed further in Section 3.6.

3.2. Burner basic performance

To investigate the effect of wood powder size on the current burner temperature profile, a total of nine samples were tested. Size distributions of nine samples, which are named from A to I, are shown in Fig. 5. The combustion stability of the wood powder burner is strongly dependent upon wood particle size. The tests on combustion stability were carried out over a short time period and are summarized in Table 4. An “Excellent” rank in this table means excellent combustion stability with sharp and stable flame as seen in the photo. A “Relatively good” means the burner optimization is expected. The shape of flame in this

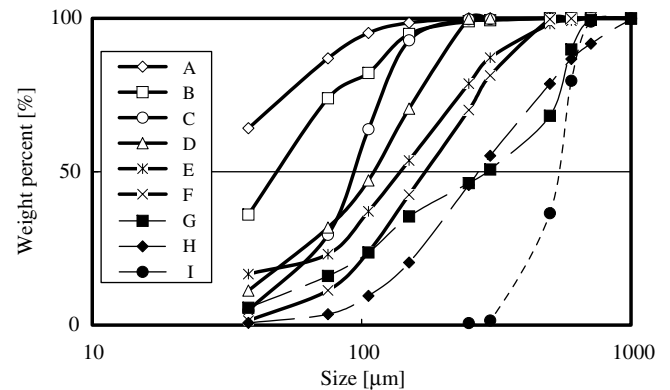


Fig. 5. Particle size distribution of wood powder tested for the combustion stability.

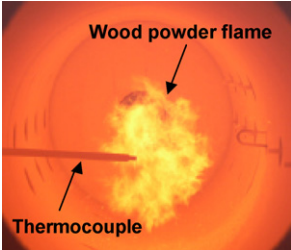
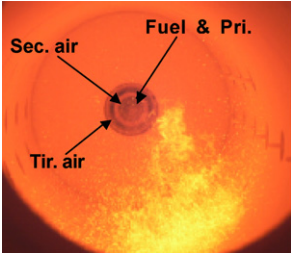
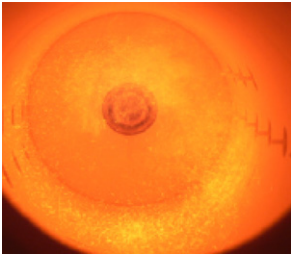
case is often indistinguishable at some times, and large particles look like dancing sparks as seen in the photo. A “Shall be improved” rank means that the burner flame cannot be seen clearly as seen in the photo. In conclusion, wood powder particles of smaller than 500 μm can be applied to the current burner without any modification. Further development and optimization of the burner will be continued in STEP-III in order to improve the burner capability and possibility for biomass use.

From visual observation, the burner flame length is strongly dependent on the air distribution as shown in Fig. 6. In cases of greater primary air with swirling, the flame length is longer. Reduced primary airflow creates a shorter flame. According to these results, the combustion chamber length can be reduced to 3 m. In the last period of STEP-II, an excellent flame and combustion performance was confirmed. As mentioned in Section 3.1, the shorter chamber design is expected to obtain better performance of the Stirling engine. In commercial design, a length of 2 m or less can be estimated.

3.3. Combustion efficiency

Measured unburned carbon concentration in fly ash and combustion efficiencies, which are calculated by Eq. (1), are shown in Fig. 7. In a higher air ratio condition, unburned carbon concentration in fly ash decreases and combustion efficiency increases. In the air ratio range between 1.1 and 1.4, unburned carbon concentrations in fly ash are 1–7%, and these are evaluated the same level as commercial PC burner. Because of the low ash contents of wood fuels comparing to normal coal fuels in Table 3, the combustion efficiency of wood powder fuel is expected to be higher than coal fuels. As shown in Fig. 7, combustion efficiencies (η_c) of wood powder calculated by Eq. (1) are excellent high level at 99.9%. As a result, unburned carbon loss of wood powder, which can be calculated as $100 - \eta_c$, is lower than 0.1% of thermal input of fuel, which can be evaluated lower than that of PC burner [25]. This result can be explained by the higher percent of volatile matter in wood powder as compared to coals. It is said that coals with

Table 4
Summary for the burning stability test results

Sample name	Top Size [μm]	D32 (SMD) ^a [μm]	Burning stability rank	Photo
A	100	59	Excellent (Combustion is stable with sharp and stable flame)	
B	250	84		
C	250	206		
D	250	145		
E	500	245		
F	500	245		
G	800	367	Relatively good (Combustion is almost stable, but dancing sparks can be seen with larger particles)	
H	1200	401		
I	800	586	To be improved (Burner flame cannot be seen clearly)	

^a D32(SDM): sautor mean diameter.

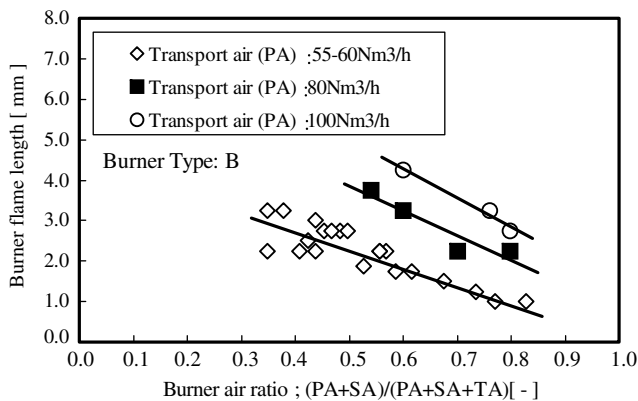


Fig. 6. Burner flame length in various air distribution.

higher volatile matter, which are sub-bituminous coal or lignite, have a tendency to decrease unburned carbon losses. And sub-bituminous coal with 50% volatile matter content, for example, is unburned carbon loss of 0.5% PC burner or around [26]. On the other hand, volatile matter content of wood powder is extremely high by 80% or more as shown in Table 3. Wood powder fuel is therefore

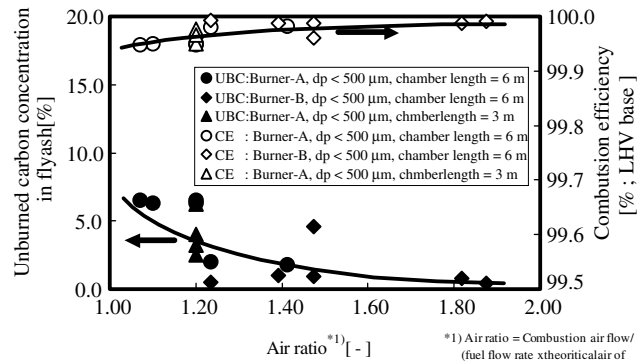


Fig. 7. Combustion efficiency in various air ratios.

expected to perform excellently on carbon burning. The combustion efficiency, η_c , depicted in Fig. 7 and discussed in the above sentence is calculated as follows:

$$\eta_c = (1 - (G_{ubc} \cdot 33.9 \cdot G_f + V_{co} \cdot 283 \cdot G_f) / (G_f \cdot LHV)) \cdot 100 \tag{1}$$

η_c combustion efficiency (%) based on lower heating value
 G_{ubc} unburned carbon content (kg/kg-fuel)

V_{co}	CO content in combustion chamber exit (kmol/kg-fuel)
G_f	fuel feed rate (kg/s)
LHV	lower heating value of fuel (MJ/kg)

Unburned carbon in fly ash was analyzed by using dust samples captured at the exit of combustion chamber, and unburned carbon content, G_{ubc} , was calculated as

$$G_{\text{ubc}} = \text{Unburned carbon in flyash (\%)/ASH}/100 \quad (2)$$

ASH Ash content in fuel (%)

3.4. CO emission

CO emission was measured at the exit of the combustion chamber using a continuous gas analyzer. CO emissions of a 10-min average value are shown in Fig. 8. When the overall air ratio is higher than 1.2, CO emission is approximately zero or less than the measurable range. When the overall air ratio is less than 1.2, CO emission increases, and combustion performance seemed to worsen. Fig. 9 shows the effect of air preheating temperature on CO emission. Cases of lower air temperature show relatively high CO emission. Therefore, a higher air ratio is required for lower air temperatures in order to maintain the same low CO emissions.

With regard to the burner and combustion system design, a lower air ratio results in better performance

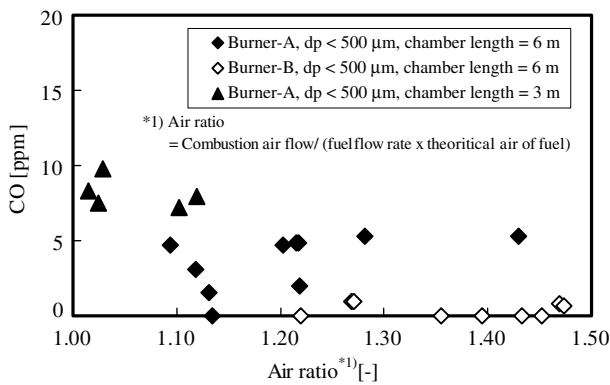


Fig. 8. CO emission in various air ratios.

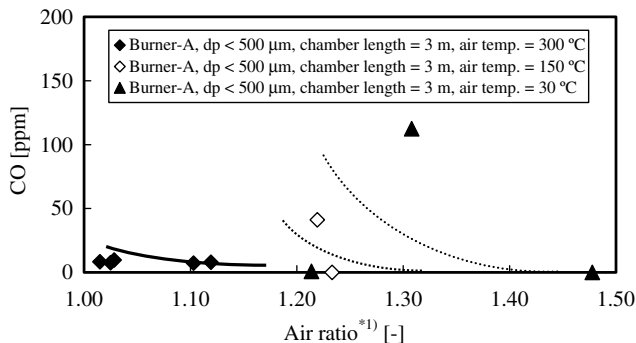


Fig. 9. Effect of preheated air temperature on CO emission.

because flue gas flow rate is lower and the capacity of the fans, blowers and flue gas treatment systems is smaller. The heat loss of flue gas is also lower. As shown in the next section on NO_x emission, a lower air ratio results in lower NO_x emission. The effect of a lower air ratio on Stirling engine performance is discussed in Section 2.6.

3.5. NO_x emission

NO_x emission was measured at the exit of the combustion chamber with an infrared (IR) continuous gas analyzer. Fig. 10 shows NO_x emission in cases of burner types A and B with combustion chamber lengths of 6 m and 3 m. NO_x emission has a good correlation with air ratio, and NO_x emission is lower than 150 ppm (in 6% O_2 conversion) in case of normal air ratio of 1.2. This is the same level as general PC boiler [25,26]. The difference in NO_x emission between burner types A and B is not so clear, but NO_x emission of chamber length of 3 m is relatively higher than that of 6 m because of the difference in gas temperature in the combustion chamber. Fig. 11 shows the effect of combustion chamber outlet gas temperature. The difference in NO_x concentration between 1000 °C

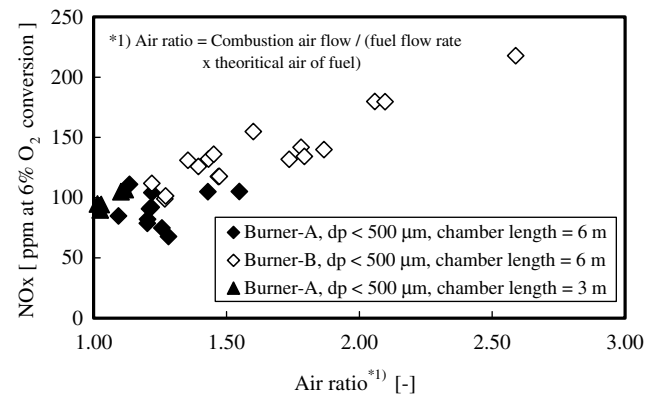


Fig. 10. Effect of burner type and length of combustion chamber on NO_x emission.

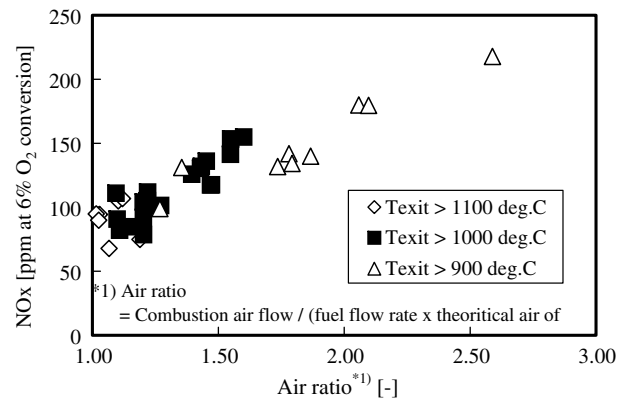


Fig. 11. Effect of the combustion chamber outlet gas temperature on NO_x emission.

and 1100 °C is not clear, but the NO_x level of 900 °C is relatively lower than that at 1000 °C.

Fig. 12 shows the effect of air distribution on NO_x emission. The abbreviation “BA” means burner air ratio defined by the sum of the primary and secondary air flow rates divided by the total combustion air flow rates. In the cases of BA > 0.4 and BA > 0.5, lower BA can reduce NO_x effectively, but in the range of BA > 0.6, NO_x reduction effect cannot be seen clearly. Lower burner air (BA) operation means better NO_x emission performance, and the possibility of reducing the burner flame length as discussed in Section 3.2. On the other hand, lower burner air means lower swirl effect, suggesting that larger particles are more difficult to burn. Optimization on the burner air ratio shall be considered from the viewpoint of burning performance of wood powder.

All of the above NO_x data show the results for an air temperature of 300 °C. The tests considering air temperature effect were carried out in a short period. Fig. 13 shows the effect of air preheating temperature on NO_x emissions. In the air ratio of 1.2–1.25, NO_x in the air temperature of 150 °C is 10–30 ppm (in 6% O₂ conversion) higher than

that of 30 °C. This is dependent on the increase of combustion temperature with the higher air preheating temperature [27]. Since air preheating temperature is typically increased up to around 600 °C in Stirling engine systems in order to achieve higher Stirling engine performance, NO_x emission property in case of the higher air preheated temperature shall be considered in more detail in the commercial stage.

3.6. Discussion on the stirling engine performance

As mentioned previously, Stirling engine performance strongly correlates to input gas temperature and gas flow rate. The higher the gas temperature and the larger the gas flow rate, the greater the power output of the Stirling engine. As shown in Fig. 4, a higher air ratio results in lower gas temperature at the combustion chamber outlet. The optimization of air ratio is needed for maximum power of the Stirling engine. Furthermore, the combustion performance and emission shall be evaluated based on the selected air ratio.

First, a simulation study for the Stirling engine power output was conducted. In this study, the Stirling engine power output is calculated with a simplified spreadsheet model. The model of the Stirling engine itself is based on the Schmidt calculation model [28]. With this model, a power output is calculated based on the internal gas temperature and pressure. And our own model for heat transfer calculation of heater tubes is attached to the above model. Finally, Stirling engine power output is modified and tuned according to our performance test results with LP gas fuel. Table 5 shows the calculation condition used for this study. A final tuning for this model will be done after completing STEP-III test. Fig. 14 shows the overall result on the power output of a 55 kW_e Stirling engine. As mentioned above, this figure clearly shows the engine power output performance correlating with entering gas temperature and entering gas flow rate.

Fig. 15 shows predicted power output of a 55 kW_e Stirling engine, which is assumed to install at the combustion chamber exit. Entering gas temperature and gas flow rate to the Stirling engine for the calculations use the plotted data shown in Fig. 4. From the results, it can be estimated that the higher air ratio results in a larger power output

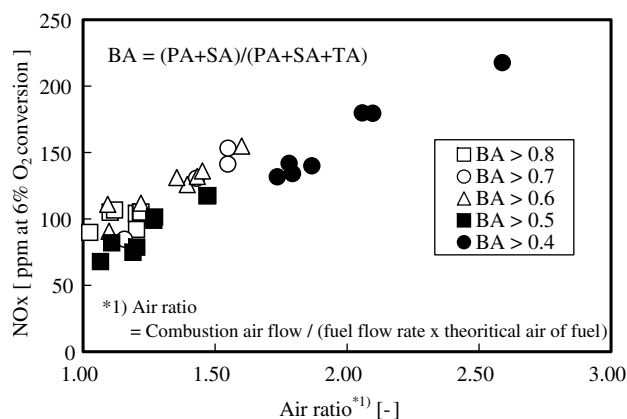


Fig. 12. Effect of the air distribution on NO_x emission.

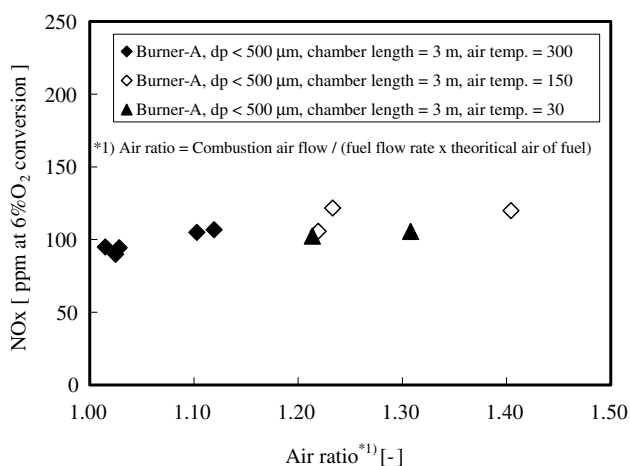


Fig. 13. Effect of air preheating temperature on NO_x emission.

Table 5
Calculation condition for 55 kW_e Stirling engine performance

Item		Calculation condition
Internal gas		Hydrogen
Internal gas pressure, average	(MPa)	15.0
Internal gas temperature, hot side	(°C)	750
Internal gas temperature, cold side	(°C)	100
Engine phase angle	(°)	90
Engine speed	(rpm)	1800
Flue gas entering flow rate	(kg/s)	(operating parameter)
Flue gas entering pressure	(kPa)	−0.10
Flue gas entering temperature	(°C)	(operating parameter)

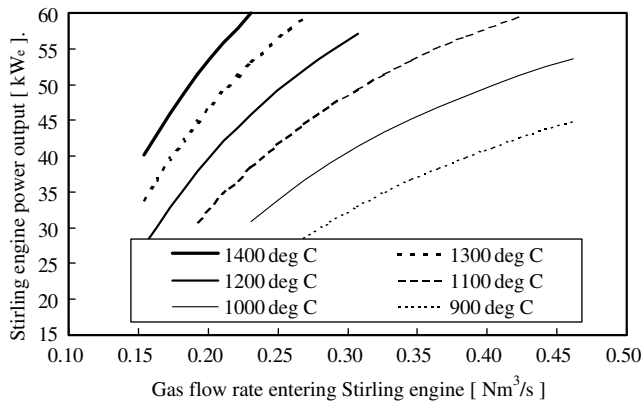


Fig. 14. Calculated 55 kW_e Stirling engine power output.

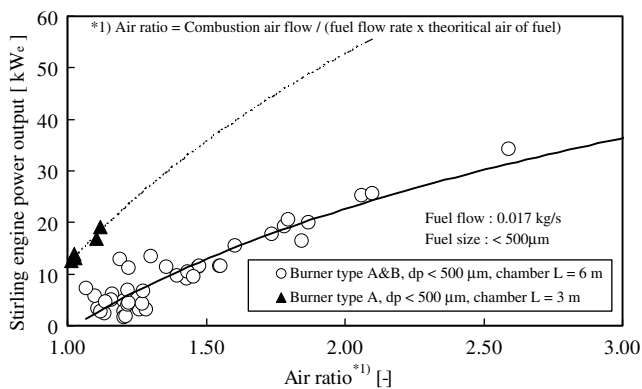


Fig. 15. Predicted power output of 55 kW_e Stirling engine installed at the combustion chamber exit.

even when the entering gas temperature is decreasing. The dotted line in Fig. 15 for the combustion chamber of 3 m suggests that an air ratio of 2.0 or larger can produce a power output of 55 kW_e for our Stirling engine.

A higher air ratio results in lower CO emission as well as higher combustion efficiency. Therefore, thermal efficiency should be greater for higher air ratios as opposed to lower air ratios. Although NO_x emission cannot be avoided in the higher air ratio operation, the concentration range can be estimated to be in the range of 150–200 ppm (at O₂ 6% conversion) according to Figs. 10–12. This range should be acceptable because the NO_x emission limitation is 350 ppm (at O₂ 6% conversion) according to the Japanese national regulations of air pollution. In case of a lower NO_x emission requirement, DeNO_x systems with or without catalysts can be easily installed in flue gas systems after the Stirling engine. This is dependent upon the customer's request.

The demonstration plant installation for STEP-III, as shown in Fig. 16 was completed at the end of March 2006. Stirling engine trial tests and its performance test with LP gas were completed as well as the trial test with wood powder burners. The demonstration tests will be continued through the end of March 2007. The above results on the Stirling engine performance will be confirmed in detail in these tests.



Fig. 16. General view of 55 kW_e Stirling engine demonstration test plant.

4. Conclusions

To develop a Stirling engine CHP system combined with simplified biomass combustion, combustion tests were carried out using different burner types. As a result of the wood powder combustion test, the basic design condition for the burner system with a 55 kW_e Stirling engine CHP system can be described as follows:

- (1) A combustion chamber length of 3 m is applicable with wood powder fuel of less than 500 μm.
- (2) The combustion performance of two different types of burners can be improved by wood powder coarser than 500 μm.
- (3) The air ratio can be reduced to 1.1 without increasing CO emission of less than 10 ppm and combustion efficiency of 99.9%. At this ratio, NO_x emission is estimated to be less than 120 ppm (6% O₂ basis).

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